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Summary

Experimental tests were performed on the OH-58A helicopter main-rotor transmission in the NASA Lewis 500-hp Helicopter Transmission Test Stand. The testing was part of a joint Navy/NASA/Army lubrication program to develop a separate lubricant for gearboxes that would improve life and load-carrying capacity. The goal of the experiments was to develop a testing procedure using a MIL–L–23699 base reference oil to fail certain transmission components, and then to run identical tests with improved lubricants and demonstrate improved performance. The tests were directed at failing components that have given the Navy problems because of marginal lubrication. These failures included mast-shaft bearing micropitting, sun gear and planet bearing fatigue, and spiral bevel gear scoring. More than 900 hr of total run time were accumulated for these tests. Some success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. Only marginal success was achieved in producing mast-shaft bearing micropitting and spiral bevel gear scoring.

Introduction

Currently, the U.S. Army uses the same lubricant for helicopter engines and transmissions. The U.S. Navy employed the same practice until 1987. The lubricant used conforms to either a MIL–L–23699 or a MIL–L–7808 specification. These oils provide satisfactory lubrication for current gas turbine engines but only marginal lubrication for transmissions. Navy overhaul depots reported increasing rejection rates of helicopter bearings and gears because of surface distress, corrosion, and wear (refs. 1 and 2). Navy AH–1T, UH–1N, and UH–1J fleets encountered upper mast-shaft bearing distress due to marginal protection capabilities of the MIL–L–23699 fluids. Early models of the Navy's SH–60B helicopters experienced spiral bevel gear surface distress when using these oils. Also, corrosion was a leading cause for rejection of transmission parts during overhaul of Navy CH–46 helicopters, with bearings accounting for most of the rejections.

In addition to marginal performance, the continued use of a common engine and transmission oil will handicap next-generation gearboxes. The potential benefits of a dedicated helicopter transmission lubricant were identified in previous studies (refs. 3 to 5). The relatively low speed and high load in certain helicopter transmission components results in extremely thin elastohydrodynamic film thicknesses between meshing gear teeth and in bearing raceway-rolling element contacts. This increases the likelihood of metal-to-metal contact and the chance of surface distress in components. Using a more viscous oil could significantly increase bearing and gear lives. For gears, tooth bending fatigue is essentially unaffected by the lubricant used; however, durability and scoring resistance are a function of the lubricant. A major effect of using an oil with improved load-carrying capacity is increased scoring and durability ratings of the gears.

The desired lubricant properties for engines are not the same as those for gearboxes. The Navy discovered that load-carrying agents added to improve gearbox durability reduced the thermal stability of the lubricant. This, in turn, decreased the fluid's performance when used in an engine. The trend for next-generation engines is toward higher operating temperatures to maximize fuel efficiency. The trend for gearboxes, however, is toward reduced weight, improved durability, increased power and load-carrying capacities, and higher temperatures. To meet the requirements for future gearboxes and address the previously discussed field problems, the Navy initiated in 1987 the Advanced Lubricants Program for helicopter transmissions.

The Naval Air Propulsion Center (NAPC) instituted a three-phase program to increase helicopter transmission life and durability through the use of advanced lubricants. The three phases of oil development are (1) interim, (2) optimum, and (3) advanced. The intent of the interim oil phase was to provide a transmission lubricant with improved load-carrying capacities to aid those gearboxes currently experiencing problems due to marginal lubrication. The interim oil was a transitional fluid between MIL–L–23699 and the optimum oil. It was compatible with MIL–L–23699 so accidental mixing would not harm turbine engines. Development of this oil allowed oil-servicing personnel an interim period to train and adjust to using a separate oil in the gearbox. Development of specification requirements for the interim oil resulted in the publication in 1986 of a new specification for a transmission oil, DOD–L–85734 (AS). The Navy implemented use of this oil in 1987. The goal of the optimum oil phase is to develop a separate lubricant for gearboxes. This lubricant should have improved load-carrying capacity (about twice that of MIL–L–23699) and improved corrosion inhibiting properties. Candidates for the optimum oil are being developed and evaluated in laboratory and bench testing by oil companies and NAPC. In the advanced oil phase, oils intended for future aircraft (at least a decade or two away) in which the transmissions will operate at high temperatures and high loads will be developed.

NASA Lewis Research Center and the U.S. Army Propulsion Directorate are supporting the Advanced Lubricants Program with their unique gear and transmission facilities. The lubricant's effect on gear pitting fatigue life is being studied using the NASA Gear Fatigue Test Apparatus at Lewis. To date, this study has found that higher viscosity lubricants (compared to MIL–L–23699) with additive packages increase gear life (ref. 6).

In addition to the gear tests, full-scale transmission tests were performed in the NASA 500-hp Helicopter Transmission Test Stand. The Army OH–58A helicopter main-rotor transmission was used for these tests. The objective of this research was twofold: first, to develop an experimental testing procedure to simulate transmission failures experienced by the Navy in the field (these lubricant-related failures include planet bearing and sun gear fatigue, mast-shaft ball bearing micropitting, and spiral bevel gear scoring); and second, to develop the testing procedure to fail the above-mentioned components in the OH–58A transmission with a MIL–L–23699 base reference oil, and then run identical tests with improved lubricants to demonstrate improved performance.

Apparatus and Test Procedure

OH–58A Main Rotor Helicopter Transmission

The OH–58A is a single-engine, land-based, light, observation helicopter. The helicopter serves both military (OH–58 Kiowa) and commercial (Bell Model 206 Jet Ranger) needs. The design maximum torque and speed for the OH–58A main-rotor transmission (fig. 1) is 350 N-m (3100 in.-lb) input torque and 6060 rpm input speed (ref. 7). This corresponds to 222 kW (298 hp). The transmission is a two-stage reduction gearbox. The first stage is a spiral bevel gear set with a 19-tooth pinion that meshes with a 71-tooth gear. Triplex ball bearings and one roller bearing support the bevel-pinion shaft. Duplex ball bearings and one roller bearing support the bevel-gear shaft in an overhung configuration.

A planetary mesh provides the second reduction stage. The bevel-gear shaft is splined to a sun gear shaft. The 27-tooth sun gear drives three 35-tooth planet gears. The planet gears mesh with a 99-tooth fixed ring gear splined to the transmission housing. The planet gears are supported by double-row spherical roller bearings attached to the planet carrier. Power is taken out through the planet carrier splined to the output mast shaft. The output shaft is supported on top by a split-inner-race ball bearing and on bottom by a roller bearing. The overall reduction ratio of the main power train is 17.44:1.

The 71-tooth bevel gear also drives a 27-tooth accessory gear. The accessory gear runs an oil pump, which supplies lubrication through jets and passageways located in the transmission housing.

NASA 500-hp Helicopter Transmission Test Stand

The OH–58A transmission was tested in the NASA Lewis 500-hp Helicopter Transmission Test Stand (fig. 2). The test stand operates on the closed-loop, or torque-regenerative, principle. Mechanical power circulates through a closed loop of gears and shafts, one of which is the test transmission. The output of the test transmission attaches to the bevel gearbox, whose output shaft passes through a hollow shaft in the closing-end gearbox and connects to the differential gearbox. The output of the differential attaches to the hollow shaft in the closing-end gearbox. The output of the closing-end gearbox connects to the input of the test transmission, thereby closing the loop.

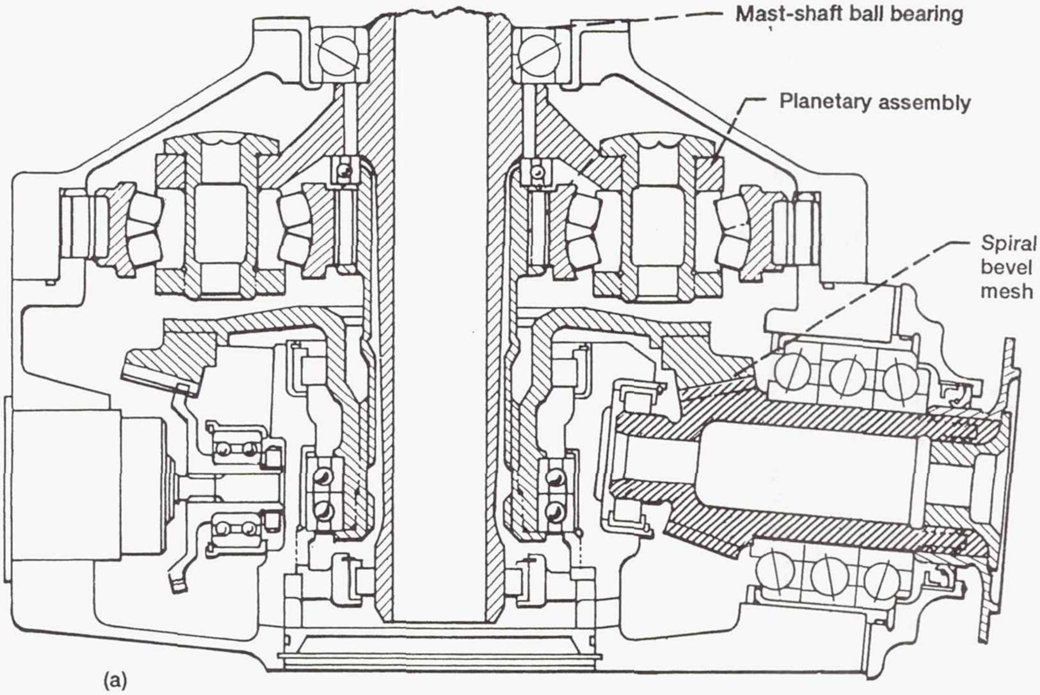
A 149-kW (200-hp) variable speed direct-current (dc) motor powers the test stand and controls the speed. The motor output attaches to the closing-end gearbox. Since power circulates around the loop, the motor replenishes only friction losses.

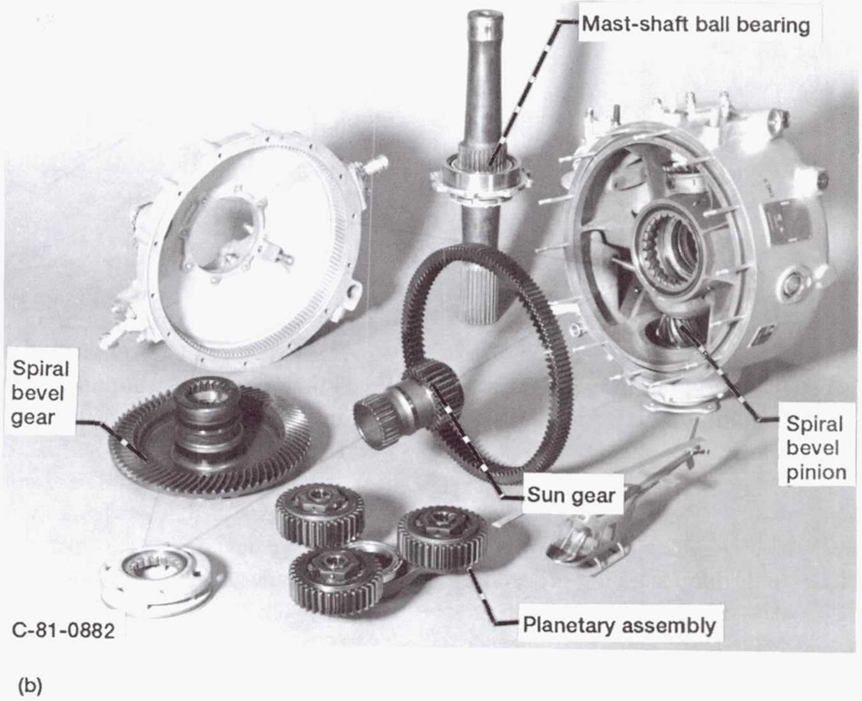
An 11-kW (15-hp) dc motor provides the torque in the closed loop. The motor drives a magnetic particle clutch. For the OH–58A application, the clutch output does not turn but exerts a torque. This torque transfers through a speed-reducer gearbox and a chain drive to a large sprocket on the differential gearbox. The torque on the sprocket puts a torque in the closed loop by displacing the gear attached to the bevel gearbox output shaft with the gear connected to the input shaft of the closing-end gearbox. This is done within the differential gearbox by a compound planetary system where the planet carrier attaches to the sprocket housing. The magnitude of torque in the loop is adjusted by changing the electric field strength of the magnetic particle clutch. For applications other than the OH–58A transmission where the speed ratio of the test transmission is slightly different or when slippage occurs (i.e., traction drives), the planet/sprocket/chain assembly rotates to make up for the speed mismatches that occur in the closed loop.

A mast-shaft loading system in the test stand simulates rotor loads imposed on the OH–58A transmission output mast shaft. The OH–58A transmission output mast shaft connects to a loading yoke. Two vertical load cylinders connected to the yoke produce lift loads. A single horizontal load cylinder connected to the yoke produces shear loads. A 13 790-kPa (2000-psig) gas nitrogen gas system powers the cylinders. Pressure regulators connected to each loading cylinder's nitrogen supply adjust the magnitude of lift and shear forces.

The test transmission input and output shafts have speed sensors, torquemeters, and sliprings. All three load cylinders on the mast yoke are mounted to load cells. The test transmission internal oil pump supplies lubrication. An external oil-water heat exchanger cools the test transmission oil.

The 149-kW (200-hp) motor has a speed sensor and a torquemeter. The magnetic particle clutch has speed sensors and thermocouples on the input and output shafts. A facility





(a) Cross section.
(b) Disassembled view.

Figure 1.—OH-58A helicopter main-rotor transmission.

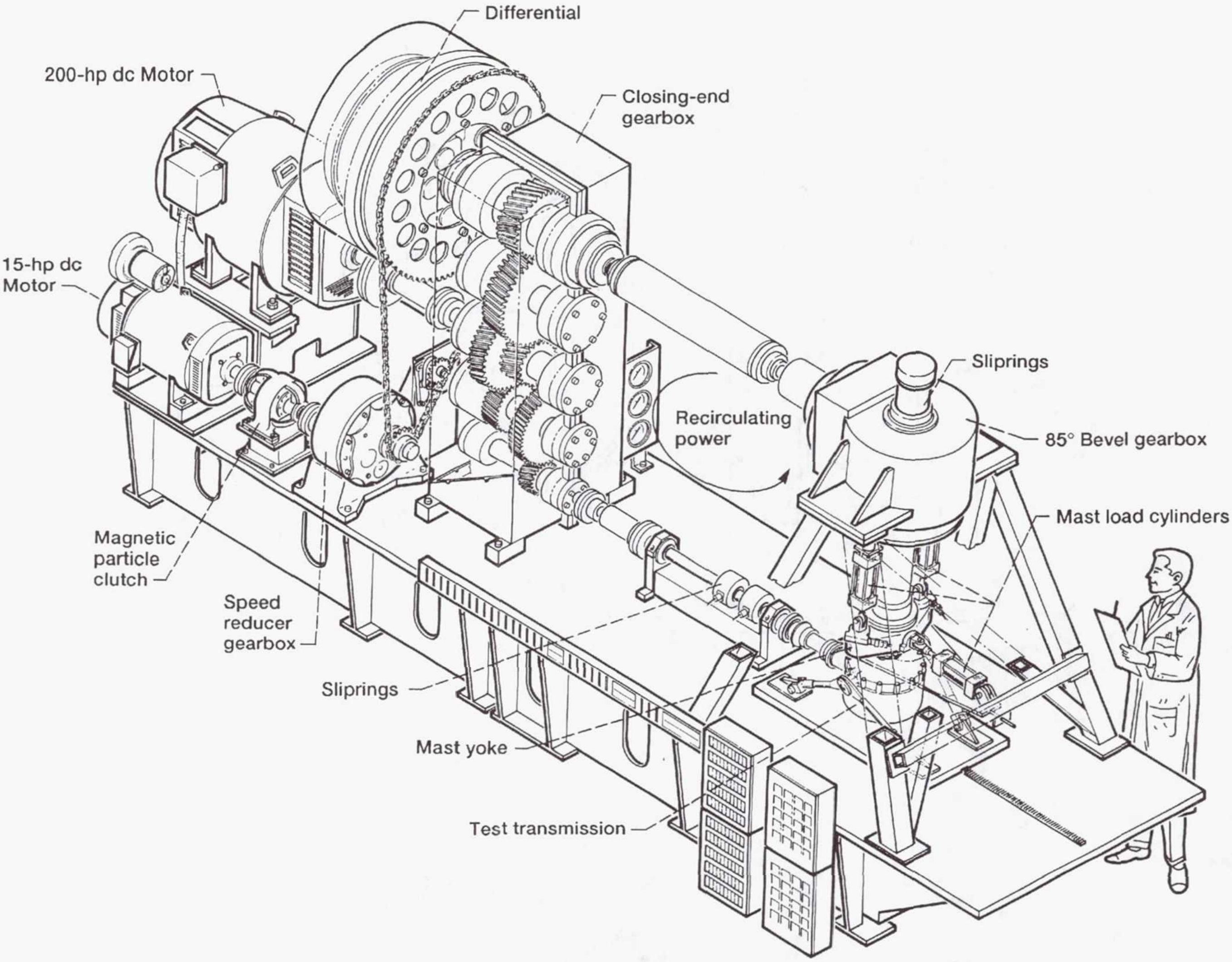


Figure 2.—NASA Lewis 500-hp Helicopter Transmission Test Stand.

oil-pumping and cooling system lubricates the differential gearbox, the closing-end gearbox, and the bevel gearbox. The facility gearboxes have accelerometers, thermocouples, and chip detectors for health and condition monitoring.

Testing Procedure

The goal of testing in the 500-hp Helicopter Transmission Test Stand was twofold: first, to develop a testing procedure to produce, on a predictable basis, component failures in the OH–58A transmission with a MIL–L–23699 base reference lubricant, and, second, to demonstrate better performance by running identical tests with improved lubricants. The component failures of interest were sun gear and planet bearing fatigue, mast-shaft ball bearing micropitting, and spiral bevel gear scoring. Ten tests were performed (table I), accumulating over 900 hr of total run time.

The transmission used for the testing was an OH–58A main-rotor transmission from the Corpus Christi Army Depot (CCAD). Before undergoing testing at Lewis, the transmission was overhauled by CCAD after it had accumulated 1908 hr of run time (load spectrum unknown). At the start of testing, all transmission components had 1908 hr on them except the spiral bevel pinion triplex ball bearing and the spiral bevel gear duplex ball bearing. These components were replaced with new ones at the overhaul. Before it was run at NASA Lewis, the transmission was disassembled, inspected, instrumented, and reassembled with the original parts. Thermocouples were installed on most bearing races. Sliprings on the transmission input and output shafts extracted the thermocouple signals from the bevel-pinion triplex and roller bearing inner races, as well as the planet bearing inner races. Accelerometers were installed at various locations on the transmission housing in a way similar to that reported in reference 8. The intent of the instrumentation was to indicate component failure during the tests.

Test 1 studied the effect of speed, load, temperature, and oil pressure on OH–58A transmission performance. The goal

TABLE I.—TEST OPERATING CONDITIONS

[Tests 2 to 10 at 6060 rpm transmission input speed, 1 percent oil-flow rate to mast ball bearing.]

Test	Time, hr	Transmission input torque, percent of design maximum	Mast radial load, percent of design maximum	Transmission oil inlet temperature		Oil type	Mast bearing ball surface roughness		Other
				°C	°F		μm AA	μin. AA	
1	---	---	---	---	---	MIL-L-23699 Brand A	---	---	Parametric studies of effect of speed, load, temperature, and oil pressure on performance
2	100	100	110	82	180		---	---	
3	50	117		121	250		---	---	
4	50	100					---	---	
5	75	117					0.53	21	
6	100						.56	22	40 percent oil-flow rate to spiral bevel mesh; four-planet system used for last 25 hr of test
7	100		132				.81	32	21 percent oil-flow rate to spiral bevel mesh; reduced oil level; four-planet system used
8	100		110			MIL-L-23699 Brand B	.81	32	21 percent oil-flow rate to spiral bevel mesh; reduced oil level
9	91		110			MIL-L-23699 Brand B	---	---	21 percent oil-flow rate to spiral bevel mesh; reduced oil level
10	100		110			DOD-L-85734	.61	24	21 percent oil-flow rate to spiral bevel mesh; reduced oil level

of this test was to combine the results of an analysis of the OH-58A transmission (appendix A) with the results from the parametric studies (appendix B) to gain insight on what operating conditions would produce the desired component failures. The transmission design maximum speed and torque were 6060 rpm input speed and 350 N-m (3100 in.-lb) input torque (ref. 7). The design maximum mast-shaft ball bearing loads were 13 334 N (3000 lb) thrust and 7934 N (1785 lb) radial (ref. 7). For test 1, transmission speed and torque were varied from 50 to 100 percent of their design maximum values. Oil inlet temperatures were varied from 82 to 135 °C (180 to 275 °F). Oil pressures were varied from 69 to 345 kPa (10 to 50 psig). Mast-shaft ball bearing loads were varied from 0 to 110 percent of their design maximum values. Except for a new mast-shaft ball bearing, all parts in the transmission used for test 1 were the original parts shipped. The lubricant used conformed to a MIL-L-23699D specification. Table II gives measured properties of the lubricant (MIL-L-23699 brand A).

Tests 2 to 7 concentrated on developing a testing procedure to fail the desired components with the MIL-L-23699 lubricant. The tests were performed sequentially, with the results of one test influencing the operating conditions of the next. The objective of test 2 was to fail the mast-shaft ball bearing. Test 2 was a 100-hr endurance run at 100-percent speed and torque, 110-percent mast shaft loading, and 82 °C (180 °F) oil inlet temperature, with the base reference MIL-L-23699 lubricant. The oil-flow rate to the mast-shaft

TABLE II.—MEASURED PROPERTIES OF MIL-L-23699 AND DOD-L-85734 LUBRICANTS

Parameter	MIL-L-23699 Brand A	MIL-L-23699 Brand B	DOD-L-85734
Kinematic viscosity, cSt			
at 38 °C (100 °F)	27.02	26.47	28.15
at 99 °C (210 °F)	5.03	5.08	5.35
Flash point, °C (°F)	260 (500)	260 (500)	254 (490)
Pour point, °C (°F)	−59 (−75)	−57 (−70)	−62 (−80)
Total acid number, mg KOH/g	0	0.37	0.40
Rubber compatibility,[a]			
H-type, swell, percent	12	161	10.5
F-type, swell, percent	13	15.3	13.1
Silicon, swell, percent	15	8.6	0
Silicon, tensile loss, percent	9	16.3	55.3
Ryder gear load capacity, percent reference oil	122	122	164
Four ball wear test, wear scar diameter, mm			
10 kg	0.569	0.251	0.239
40 kg	0.867	0.424	0.394

[a]For silicon rubber compatibility tests, MIL-L-23699 run at 121 °C, DOD-L-85734 run at 110 °C.

ball bearing was reduced to 1 percent of its nominal value. This was accomplished by soldering closed the oil jet that lubricated the bearing and drilling a new hole that was approximately one-quarter of the original diameter. The flow rates were measured before and after modification by connecting the jet to an external oil-pumping system and measuring the collected oil after a given amount of time. All transmission parts used for test 2 were the same as in test 1. Any part which failed during test 2 was replaced with a new component.

The objective of test 3 was to fail the mast-shaft ball bearing, fatigue the sun and planet gears and bearings, and score the spiral bevel gears. Test 3 was a 50-hr endurance run with conditions similar to test 2 but with higher power (117-percent torque) and a higher oil inlet temperature (121 °C (250 °F)). Based on the OH-58A analysis (appendix A), 117-percent torque was the highest torque at which it was believed safe to operate the transmission to promote gear pitting fatigue while avoiding tooth bending fatigue.

The objective of test 4 was the same as that of test 3. Test 4 was conducted at 100-percent torque and 121 °C oil inlet temperature to determine if the results of test 3 could be duplicated. The oil-flow rate to the mast-shaft ball bearing for tests 3 and 4 (and all subsequent tests) was, as in test 2, 1 percent of the nominal. Again, any part that failed during testing was replaced with a new component.

The objective of test 5 was to repeat the failures of test 3 (sun gear and spiral bevel mesh failures; see **Results and Discussion**) and fail the mast-shaft ball bearing. Test 5 was a 75-hr endurance run with the torque and temperature the same as in test 3. The balls of the mast-shaft ball bearing were modified to decrease the elastohydrodynamic film thickness-to-surface roughness ratio. The balls' surfaces were roughened by placing one ball at a time in a glass container, with a mixture of 200 g of 20 grit silicon carbide. The remainder of the jar was filled with 6.35-mm (0.25-in.) steel balls. The container was tumbled at 300 rpm for 6 hr. This was repeated for each ball of the bearing. The design specifications for the maximum surface roughness for a new bearing ball was 0.05 μm arithmetic average (AA) (2 μin. AA). The goal was to roughen the balls to about 0.51 μm AA (20 μin. AA). The surface roughness of the balls after tumbling was measured using a Tallysurf measuring system. For the mast-shaft ball bearing of test 5, the average surface roughness of the balls was 0.53 μm AA (21 μin. AA).

The objective of test 6 was to reproduce the failures of tests 3 and 5 (sun gear, planet bearing, spiral bevel mesh, and mast-shaft ball bearing failures; see **Results and Discussion**). Test 6 was a 100-hr endurance run at 117-percent torque and 121 °C oil inlet temperature. A new mast-shaft ball bearing was installed at the start of test 6. Before installation, the bearing balls were tumbled as in test 5. The average surface roughness of the balls was 0.56 μm AA (22 μin. AA). For the last 25 hr of the test, the oil jet to the spiral bevel mesh was modified to reduce the oil flow and promote tooth scoring. The jet was modified by plugging one of the two oil-jet holes lubricating the mesh. The modified oil-flow rate, as measured with an external oil-pumping system, was 40 percent of the nominal. Also for the last 25 hr of the test, a four-planet OH-58C assembly was used along with the modified bevel mesh oil jet to focus on spiral bevel gear scoring rather than planetary failures. In addition to having one more planet, the four-planet assembly is different from the three-planet assembly in that the planet bearings are cylindrical rollers rather than spherical, the planets are straddle-mounted by the carrier rather than overhung, and the four-planet assembly has significantly higher load carrying capacity.

The objective of test 7 was to concentrate on scoring the spiral bevel gears and to micropit the mast-shaft ball bearing. Test 7 was a 100-hr endurance run at 117-percent torque and 121 °C oil inlet temperature. The oil jet for the spiral bevel mesh was further modified by soldering the holes closed and redrilling one of the two jet holes to approximately one-half the original size. The jets were originally designed to spray oil to a given target area. This original design ensured that the jets sprayed oil to the gear mesh when installed in the transmission. A mounting fixture was fabricated to measure the flow rate and to ensure that the oil-jet target was within the design specifications. The modified oil-flow rate, as measured by an external oil-pumping system, was 21 percent of the nominal. Additionally, the transmission was operated with less oil to guarantee that the spiral bevel pinion was not submerged in any oil during running. A four-planet assembly was used in test 7. A new mast-shaft ball bearing was installed at the start of test 7. Before installation, the bearing balls were tumbled as in test 5. The average surface roughness of the balls was 0.81 μm AA (32 μin. AA). For this test, the bearing was run at 132 percent of the design maximum radial load.

Tests 8 and 9 concentrated on repeating the results of tests 2 to 7, but with a second, commercially available brand of MIL–L–23699 lubricant (see table II for measured properties of MIL–L–23699 brand B). Tests 8 and 9 were a combination of tests 5 to 7. They were planned 100-hr endurance runs at 100-percent speed, 117-percent torque, 110-percent mast-shaft loading, and 121 °C oil inlet temperature. For test 8, the same mast-shaft ball bearing from test 7 was used. For test 9, a new mast-shaft ball bearing was used but the balls were not tumbled before running. The modified jet to the spiral bevel mesh with 21-percent oil-flow rate was used along with the reduced oil level. Additionally, the three-planet system was reinstalled with new planets and sun gears at the start of both tests. Once again, any component that failed during tests 8 and 9 was replaced with a new one.

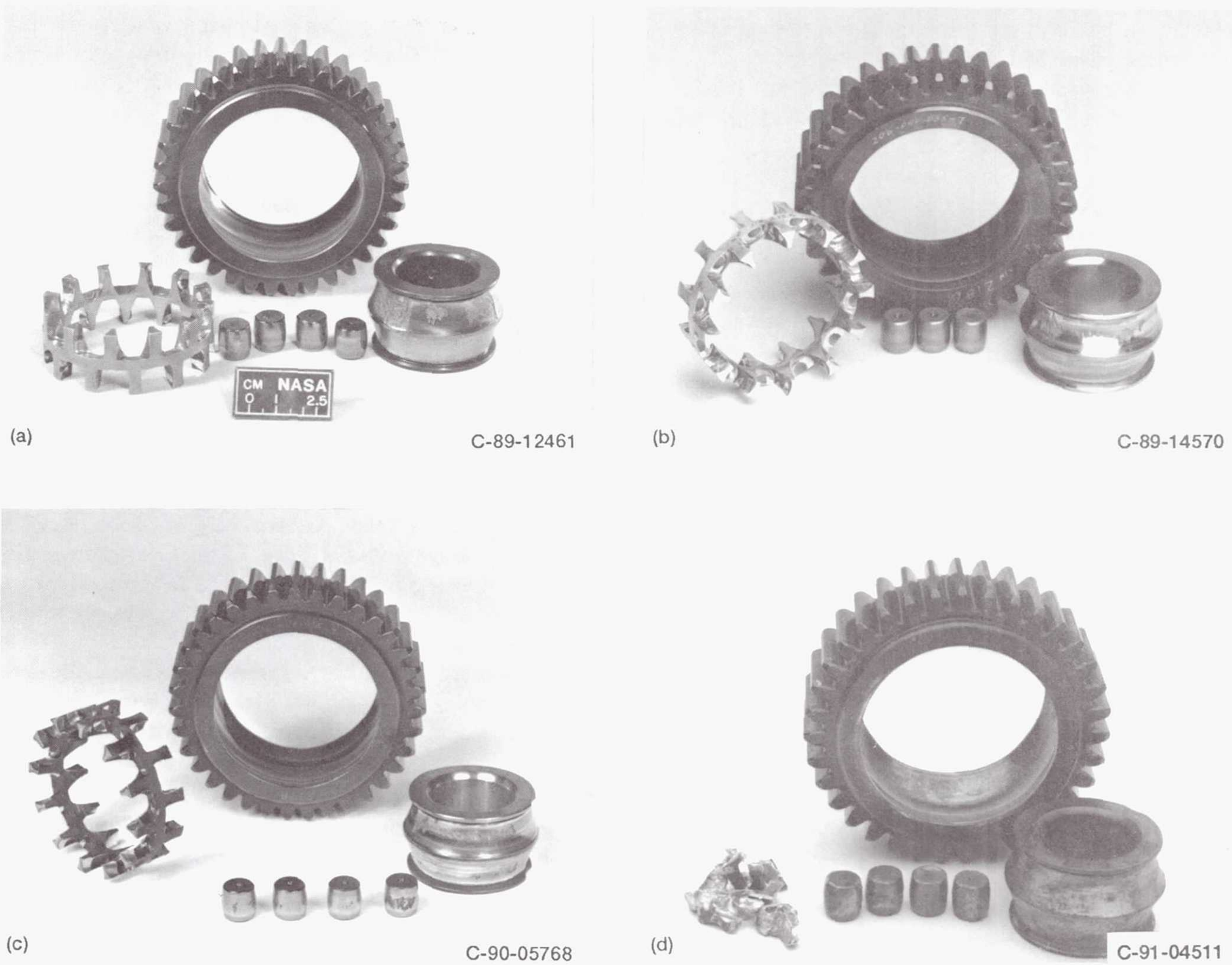
Test 10 was a repeat of test 8 but using a DOD–L–85734 lubricant. A new mast-shaft ball bearing was used. The balls were tumbled prior to running, and the average ball roughness measured was 0.61 μm AA (24 μin. AA). Table II gives measured properties of the DOD–L–85734 lubricant. The physical and chemical properties of the MIL–L–23699 and DOD–L–85734 lubricants (viscosity, flash and pour points, acidity, etc.) are similar. The DOD-L-85734 lubricant has significantly higher load-carrying capacity as indicated by the Ryder Gear Ratings (and somewhat by the Four Ball Wear tests). One consequence of the increased load-carrying additives is reduced rubber compatibility as indicated by the silicon tests.

Results and Discussion

Planetary Results

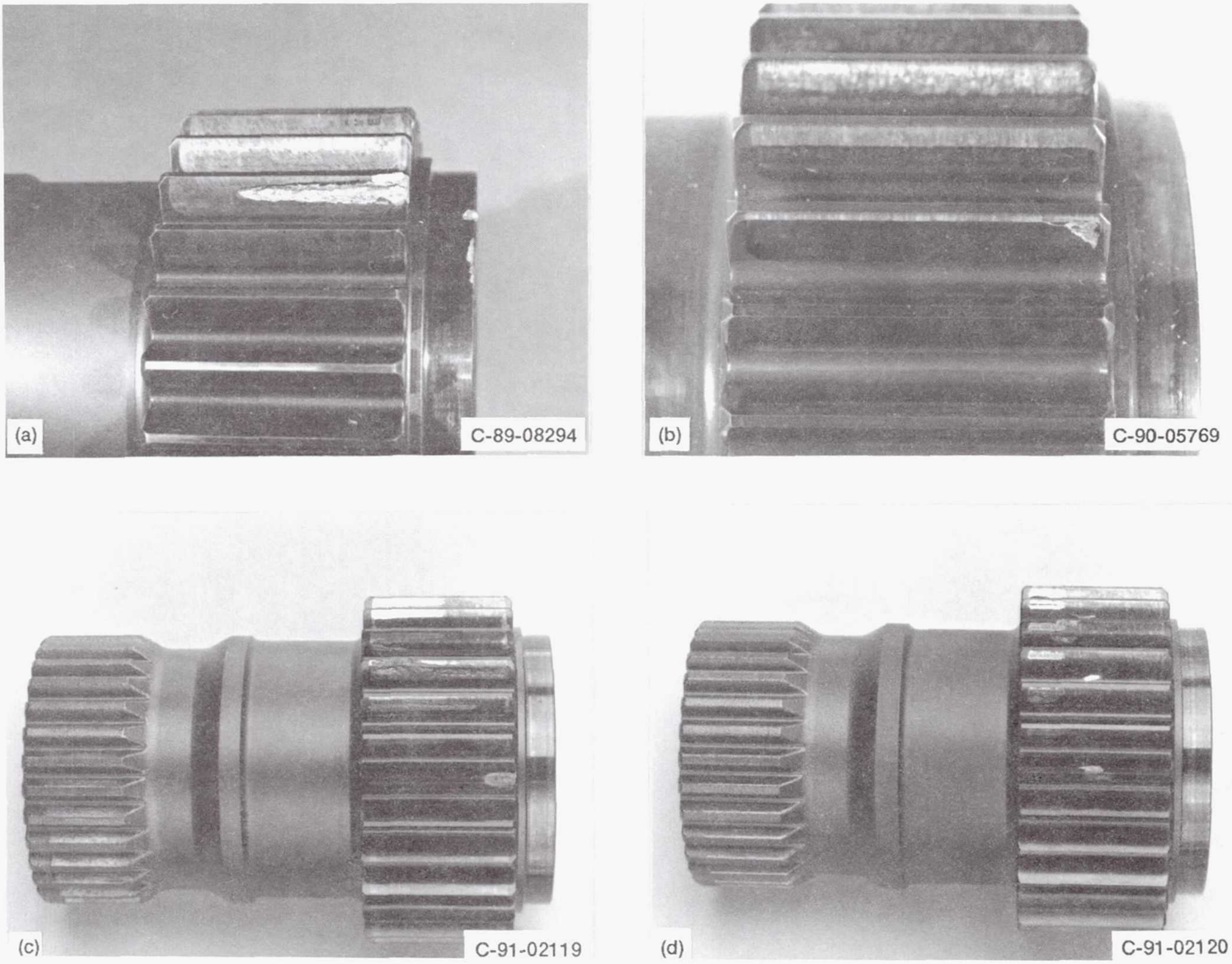
Some success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. A total of three planet bearing fatigue failures and one planet bearing cage failure were produced with MIL-L-23699 lubricants (fig. 3). A total of four sun gear fatigue failures were produced with MIL-L-23699 lubricants (fig. 4).

Planet bearing failures.—For MIL-L-23699 lubricant brand A, three planet bearing fatigue failures occurred. The first occurred on one of the original planet gears of the transmission during test 5 (serial number (S/N) B12-25780, fig. 3(a)). The failure was a fatigue spall on the inner race. The component acquired 1908 hr of run time (load spectrum unknown) before NASA testing, 71 hr in NASA test 1, 100 hr in test 2, 50 hr in test 3, 50 hr in test 4, and 16 hr in test 5. The 66 hr of run time from tests 3 and 5 were at operating conditions of 117-percent torque and 121 °C (250 °F) oil inlet temperature. It was assumed no overload occurred during the first 1908 fleet hours. On two other planet bearings, inner-race fatigue failures occurred at 117-percent torque and 121 °C oil inlet temperature. One occurred after 59 hr of test 5



(a) Serial number (S/N) B12-25780, fatigue failure during test 5.
(b) S/N B12-35194, fatigue failure after test 5.
(c) S/N B12-35181, fatigue failure after test 6.
(d) S/N B12-25580, cage failure after test 9.

Figure 3.—OH-58A transmission planet bearing failures.



(a) S/N B12-3993, fatigue failure after test 3.
(b) S/N B12-00566, fatigue failure after test 6.
(c) S/N B12-3162, fatigue failure and scoring during test 8.
(d) S/N B12-4070, fatigue failure and scoring after test 8.

Figure 4.—OH-58A transmission sun gear failures.

(S/N B12-35194, fig. 3(b)) and the other after 75 hr of test 6 (S/N B12-35181, fig. 3(c)). All three failures were similar: the spalls covered both rows of each bearing and about one-third of each circumference.

For MIL-L-23699 lubricant brand B, one planet bearing cage failure was produced after 91 hr of test 9 (S/N B12-25580, fig. 3(d)). This failure was catastrophic; the cage for one of the planet bearings was destroyed and the rollers and raceways showed excessive scoring and distress.

Sun gear failures.—Two sun gear fatigue failures were produced with MIL-L-23699 lubricant brand A. The first occurred on the transmission's original sun gear after test 3 (S/N B12-3993, fig. 4(a)). A large fatigue spall covering nearly three-quarters of the face width and much of the tooth load zone occurred on one tooth. The component had acquired 1908 hr of run time (load spectrum unknown) before NASA testing, 71 hr in NASA test 1, 100 hr in test 2, and 50 hr in test 3. The second sun gear failure, a small pit at the edge of one tooth, occurred after test 6 (S/N A12-00566, fig. 4(b)). The gear had acquired 50 hr in test 4, 75 hr in test 5, and 100 hr in test 6 (75 hr with a three-planet mesh and 25 hr with a four-planet).

Two additional sun gears experienced fatigue failures using the MIL-L-23699 lubricant brand B. The first was during test 8 (S/N B12-3162, fig. 4(c)). This gear had spalls and scoring on many of its teeth. This gear acquired 100 hr of test 7 (117-percent torque, 121 °C, MIL-L-23699 lubricant brand A) and 80 hr in test 8 (117-percent torque, 121 °C, MIL-L-23699 lubricant brand B). The final sun gear failure occurred after running the last 20 hr of test 8 (S/N B12-4070, fig. 4(d)). This gear also had spalls and scoring on many of its teeth.

Additional comments.—The results indicate that the testing procedure with operating conditions of 117-percent torque and 121 °C oil inlet temperature produced planetary fatigue failures repeatably. Traditionally, some scatter occurs in fatigue lives for gear and bearing components run under identical conditions. Because of the relatively low number of failure data points (compared to conventional component fatigue testing), the confidence level for reproducing failures is not high. However, the intent of the testing was not to replace gear and bearing component fatigue testing, but to produce certain types of failures in a real transmission with its complex interactions of gears, bearings, and housing. From this viewpoint, success was achieved. A procedure was developed to produce planetary fatigue failures.

There was a problem with repeatability. Tests run at 117-percent torque and 121 °C oil inlet with the MIL-L-23699 lubricant brand A produced planet bearing inner-race fatigue failures within about 75 hr. Under these conditions, no bearing fatigue failures occurred in two successive 100-hr tests with the MIL-L-23699 lubricant brand B. It is not certain if the difference in results was due to oil differences. Sun gear fatigue failures occurred within about 180 hr at 117-percent torque and 121 °C oil inlet temperature. The sun gear results were slightly ambiguous since components were run at different combinations of operating conditions. The operating conditions were varied because the tests were performed sequentially, with the results from one test influencing the conditions of the next.

A single 100-hr run using a DOD-L-85734 lubricant (test 10) produced no component failures. Although this was encouraging, because of the limited amount of data, these results show that more testing would be necessary to demonstrate the fatigue life of the planetary system using a DOD-L-85734 lubricant is greater than that using a MIL-L-23699.

Mast-Shaft Ball Bearing Results

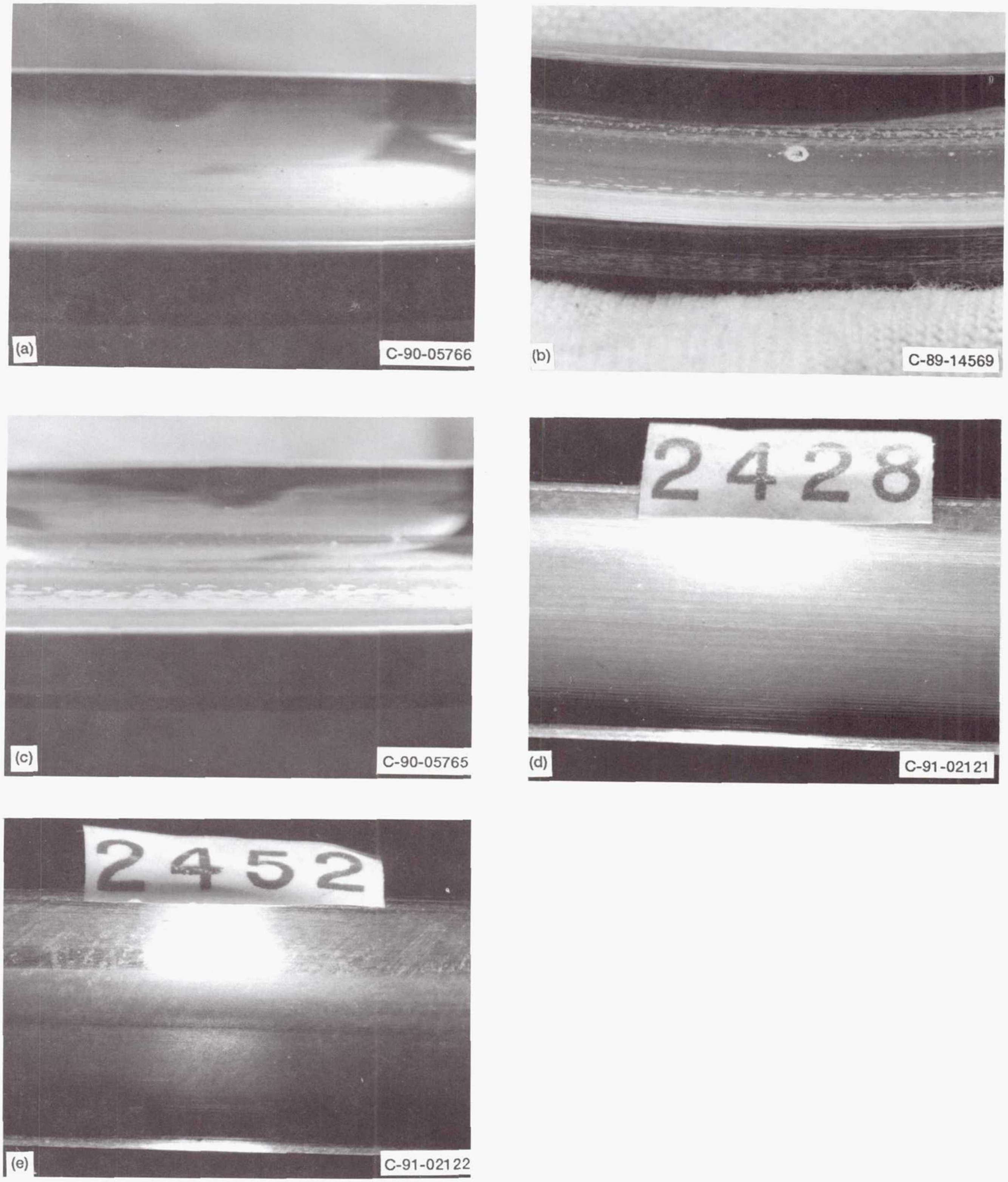
Only slight success was achieved in developing a testing procedure to produce micropitting failures in mast-shaft ball bearings. Six bearings were tested, but micropitting occurred in only one (fig. 5). The first bearing tested acquired 71 hr in test 1, 100 hr in test 2, 50 hr in test 3, and 50 hr in test 4. All tests were run at 110-percent design maximum radial load and 101-percent thrust load. After these tests, the bearing was in very good condition, with no significant wear marks on the balls or raceways (S/N 2522, fig. 5(a)). Reducing the oil-flow rate to 1 percent of the nominal had no effect on the results.

A second bearing was modified by tumbling the balls to increase their surface roughness previously described. After the balls were tumbled and before the test was run, the surface roughness of each ball was measured. Four measurements were taken for each of the 14 balls (fig. 6). Scatter in the measured roughness existed not only among the various balls but also in the measurements for each ball. The average surface roughness of the balls was 0.53 ± 0.10 μm AA (21 ± 4 μin. AA) where 0.10 μm (4 μin.) was one standard deviation. The bearing was assembled in the transmission, which was run 75 hr at 110-percent design maximum radial load and 121 °C (250 °F) oil inlet temperature (test 5). After running, the bearing exhibited micropitting on the inner race (S/N 2512, fig. 5(b)). The surface roughness of each ball was remeasured (fig. 6). The average surface roughness was reduced to 0.09 ± 0.02 μm AA (3.6 ± 0.7 μin. AA). Thus, the induced ball roughness was reduced substantially after running 75 hr.

This testing procedure was repeated with a third bearing. The average ball surface roughness of the balls after tumbling was 0.56 ± 0.13 μm AA (22 ± 5 μin. AA). The bearing was run 100 hr (test 6) at the same operating conditions as was the previous bearing that micropitted. After the run, the bearing had a noticeable wear mark along the load track of the inner race, but micropitting was not evident (S/N 2472, fig. 5(c)). Each ball's surface roughness was remeasured, and the average value was 0.09 ± 0.02 μm AA (3.6 ± 0.7 μin. AA). Thus, the average ball surface roughness of the balls both before and after running was nearly identical for tests 5 and 6. However, micropitting was not produced in test 6.

The balls of a fourth bearing were tumbled, and the transmission was run as previously described. The average surface roughness after tumbling was 0.81 ± 0.13 μm AA (32 ± 5 μin. AA). This was higher than the average for the previous bearings even though the tumbling procedure was the same. The bearing was run 100-hr at 132-percent design maximum radial load (test 7). As in test 6, the bearing had a noticeable wear mark along the load track of the inner race, but micropitting was not evident. The same bearing was used for test 8 (100 hr at 110-percent design maximum radial load and MIL-L-23699 lubricant brand B). The bearing still exhibited no signs of micropitting after testing (S/N 2428, fig. 5(d)).

The fifth bearing tested was run for 91 hr with MIL-L-23699 lubricant brand B (test 9). The bearing balls were not tumbled prior to running. After running, the bearing was in very good condition and looked similar to the first bearing tested (S/N 2522, fig. 5(a)). The last bearing tested was run 100 hr with the DOD-L-85734 lubricant (test 10). The bearing balls were tumbled prior to running, and the average surface roughness of the balls after tumbling was 0.61 ± 0.15 μm AA (24 ± 6 μin. AA). After running, the bearing looked similar to those of tests 5 to 7 (S/N 2452, fig. 5(e)). In summary, the two bearings that had no surface modifications to the balls were in good condition after running with no significant wear on the balls or raceways. Of the four bearings with balls that had surface modifications, one exhibited micropitting. The remaining three had heavy wear tracks on the inner races but no micropitting.



(a) S/N 2522 after tests 1 to 4, balls not tumbled.
(b) S/N 2512, micropitting after test 5.
(c) S/N 2472 after test 6.
(d) S/N 2428 after tests 7 and 8.
(e) S/N 2452 after test 10.

Figure 5.—OH-58A transmission mast-shaft ball bearing inner race photos.

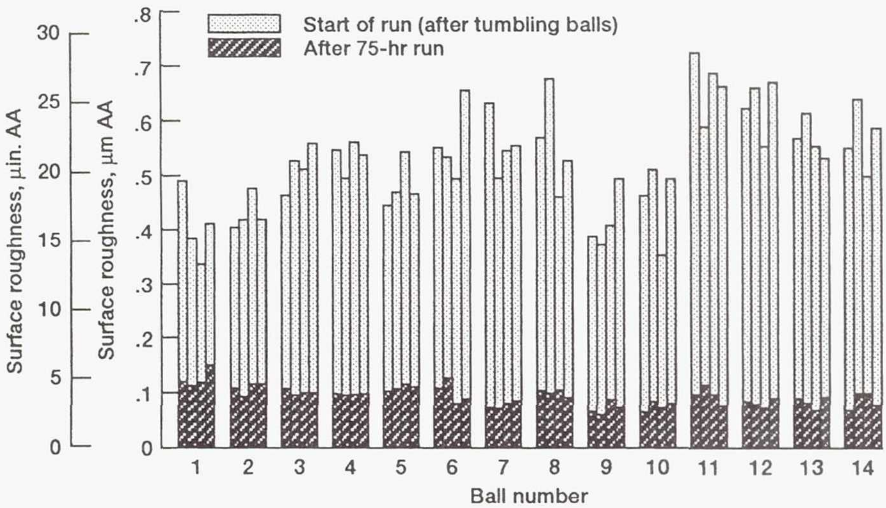


Figure 6.—Surface roughness measurements of balls for OH-58A transmission mast-shaft ball bearing, S/N 2512.

Spiral Bevel Mesh Results

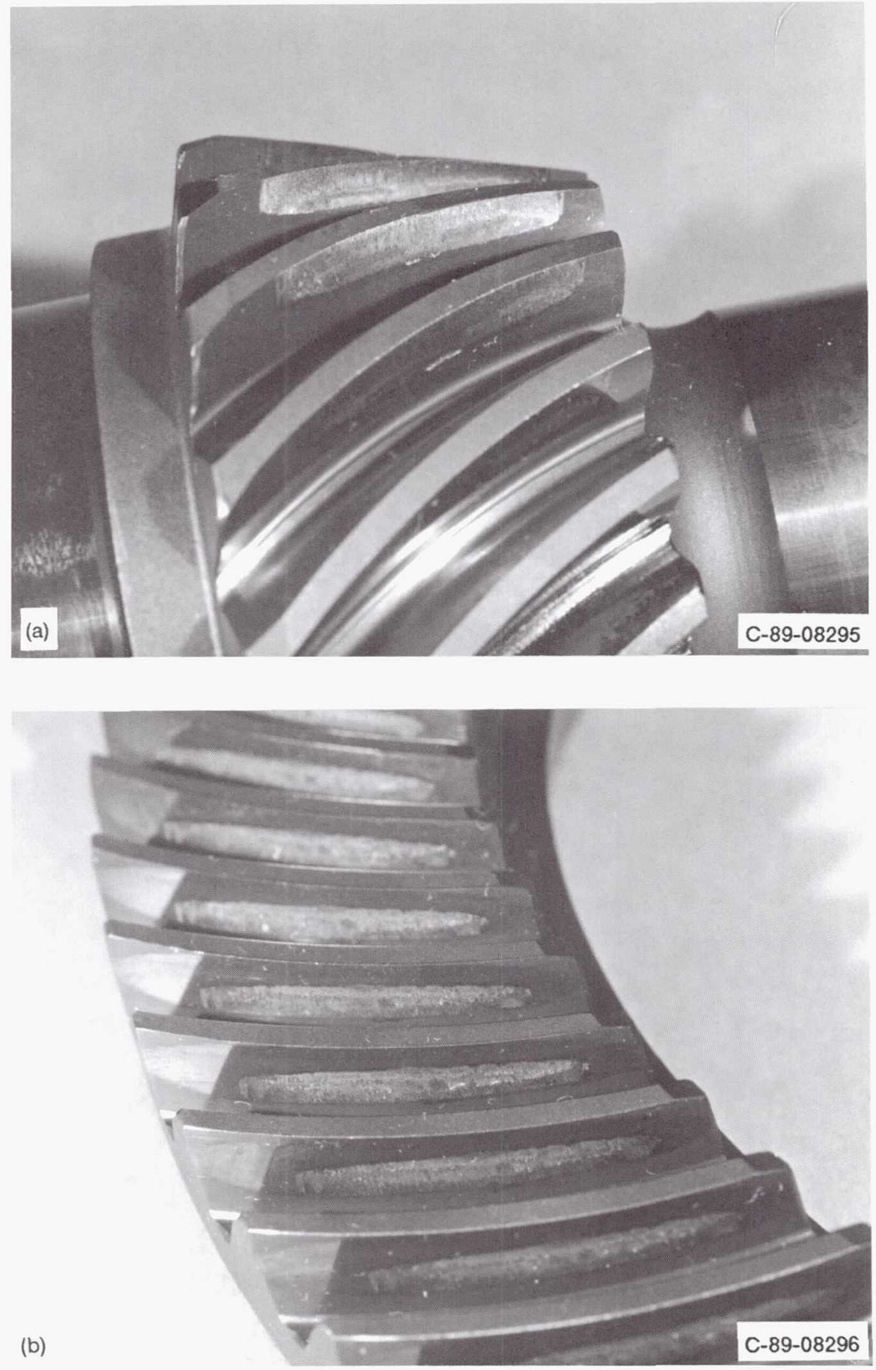
Only slight success was achieved in developing a testing procedure to produce spiral bevel gear scoring failures. The test of the first set of bevel gears was unsuccessful in that a fatigue spall was produced on one pinion tooth during the middle of test 2. Other than the spall, both the pinion and the gear showed good contact pattern and no signs of surface distress. These were the transmission's original bevel gears, with 1908 hr in run time before NASA testing, 71 hr in NASA test 1, and 51 hr in test 2.

The second set of bevel gears tested showed signs of heavy wear and light scoring after test 3 (fig. 7). This set was run 56 hr in test 2 (100-percent torque and 82 °C (180 °F) oil inlet temperature) and 50 hr in test 3 (117-percent torque and 121 °C (250 °F) oil inlet temperature). After test 2, the set showed good contact pattern and no surface distress. After test 3 the set exhibited heavy wear, light scoring, and some micropitting. Heavy wear was evident on the pinion from the root to the tip along most of its face width (fig. 7(a)). The pinion also had small pit lines about 6 to 9 mm wide in the middle of the tooth face and toward the root at the start of contact. The output gear exhibited heavy wear in the middle of the tooth along most of its face width (fig. 7(b)). A high contact pattern that developed on the pinion (pattern toward the tooth tip) caused some concern. The reason for this high pattern is not known.

A third set of bevel gears was tested in an attempt to repeat this failure mode. This set of gears was initially run 50 hr at 100-percent torque and 121 °C oil inlet temperature (test 4) to determine if high temperature alone would produce scoring. After running, the gear set was inspected; the contact pattern was good and no tooth distress had been produced. The mesh was then run 75 hr at 117-percent torque and 121 °C oil inlet temperature (test 5). Still no tooth distress was evident. The mesh was run an additional 75 hr at these conditions (test 6). Again, no tooth distress was evident.

The lubricant flow rate to the bevel mesh was modified in an attempt to promote tooth scoring. After the transmission was running 25 hr at 40-percent oil-flow rate (the last 25 hr of test 6) no signs of tooth distress were evident. The mesh was then run 100 hr at 21-percent oil-flow rate and reduced oil level (test 7). After the run, both the pinion and gear exhibited a thin, elliptical wear mark but no scoring. The wear marks were on the pinion near the root at the start of contact and on the gear near the tip, both extending the majority of the tooth widths. The gears were run an additional 100 hr at these conditions with MIL-L-23699 lubricant brand B (test 8). No change in the gears was apparent after this test.

A fourth and fifth set of bevel gears were tested at 117-percent torque, 121 °C oil inlet temperature, 21-percent oil-flow rate, and reduced oil level with MIL-L-23699 lubricant brand B and DOD-L-85734 lubricant (tests 9 and 10). The gears showed no evidence of tooth scoring or distress after these tests. Thus, a testing procedure to repeatedly produce spiral bevel gear scoring was not achieved.



(a) Pinion.
(b) Output gear.

Figure 7.—OH-58A transmission spiral bevel mesh after test 3.

Summary of Results

Experimental tests were performed on the OH-58A helicopter main-rotor transmission in the NASA 500-hp Helicopter Transmission Test Stand. The objective was to develop a procedure to produce sun gear and planet bearing fatigue failures, mast-shaft ball bearing micropitting, and spiral bevel gear scoring. These components were typical of those in which the Navy found problems due to marginal lubrication. The following results were obtained:

1. Success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. Three planet bearing fatigue failures occurred with a MIL–L–23699 lubricant. Each failure occurred within 75 hr at 117-percent design maximum torque and 121 °C (250 °F) oil inlet temperature. Four sun gear fatigue failures occurred with two different brands of MIL–L–23699 lubricant. Some repeatability problems existed.

2. Only slight success was achieved in developing a testing procedure to produce mast-shaft ball bearing micropitting. Four bearings had modifications performed to roughen ball surface finishes and promote marginal lubrication. Of these four, one bearing exhibited micropitting when run in endurance tests.

3. A testing procedure to repeatedly produce spiral bevel gear scoring was not achieved. One set of gears out of five exhibited heavy wear and light scoring when run at 117-percent torque and 121 °C oil inlet temperature. Reducing the oil-flow rate to 21 percent of nominal had little effect on tooth surface distress.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, December 11, 1991

Appendix A
OH–58A Transmission Analysis

The component failures desired as part of the testing in the 500-hp Helicopter Transmission Test Stand were planet bearing and sun gear fatigue, mast-shaft ball bearing micropitting, and spiral bevel gear scoring. Gear or bearing fatigue is the failure mode resulting from the repeated application of loads. Pitting or spalling occurs when the fatigue limit of a material is reached. Surface pitting and micropitting usually result from localized load concentrations and low lubrication film thickness between contacting surfaces. Spalling is characterized by large particles or chips which spall or flake off of a surface. It occurs after smaller pits have developed or when much sliding occurs between surfaces. Pitting and spalling are functions of surface geometry, loads, number of load cycles, material properties, surface finishes, and lubrication properties.

Gear tooth scoring is a severe wear condition in which surfaces weld together as a result of metal-to-metal contact and then tear apart. Scoring is not a fatigue phenomenon and can happen whenever appropriate conditions exist. The basic mechanism of scoring is caused by intense frictional heat generated by high sliding, high surface pressure, and high temperature. Scoring is a function of surface geometry, loads, surface finish, speed, temperature, and lubrication properties.

An analysis was performed on the OH–58A main-rotor transmission. Predictions of gear bending and contact stress, spiral bevel gear scoring, planetary gear and bearing performance, and a mast-shaft ball bearing performance were performed.

AGMA Stresses and Scoring Analysis

A bending and contact stress analysis using the American Gear Manufacturers Association (AGMA) standards was performed on the OH–58A transmission main power train. A scoring analysis was also performed on the spiral bevel gear mesh. Table III summarizes the stresses. The AGMA bending and contact stresses at 222 kW (298 hp) were taken from the OH–58A transmission design report (ref. 7). (The bending stresses in the design report were based on methods from refs. 9 and 10 and the contact stresses were based on refs. 11 and 12.) AGMA bending stresses are a linear function of gear tooth tangential load. Contact stresses are a function of the square root of the load. Thus, the bending and contact stresses at 262 kW (350 hp) in table III were based on the values at 222 kW and scaled proportionally. The assumed allowable AGMA bending and contact stresses in table III were from references 9 to 13 and indicate a range of the design limits.

As expected, all values of bending and contact stresses at 222 kW (100-percent design load) fell within or below the assumed allowable range. At 262 kW, all component stresses except those of the planet gear fell within or below the assumed allowable range for bending stresses. The planet gear stresses were higher than the allowables because of the reduction in allowable stress due to reversed bending (70-percent of nominal). If this reduction was not considered, the planet gear would have been within the allowable stress range. The ring

TABLE III.—OH–58A TRANSMISSION GEAR STRESSES

Component	Stress at 100-percent torque,[a] 222 kW at 6060 rpm		Stress at 117-percent torque, 262 kW at 6060 rpm		Assumed allowable stress[b]	
	MPa	ksi	MPa	ksi	MPa	ksi
Bending stress						
Spiral bevel pinion	186	27	219	32	380 to 485	55 to 70
Spiral bevel gear	165	24	194	28	380 to 485	55 to 70
Sun gear	276	40	324	47	380 to 450	55 to 65
Planet gear (mesh with sun gear)	310	45	364	53	269 to 317[c]	39 to 46
Planet gear (mesh with ring gear)	290	42	340	49	269 to 317[c]	39 to 46
Ring gear	372	54	437	63	380 to 450	55 to 65
Contact stress						
Spiral bevel mesh	1434	208	1554	225	1380 to 1550	200 to 225
Sun-planet gear mesh	1248	181	1352	196	1380 to 1550	200 to 225
Planet-ring gear mesh	1151	167	1248	181	1380 to 1550	200 to 225

[a]From reference 17.
[b]From references 9 to 13.
[c]Reduced to 70 percent of nominal due to reversed bending.

TABLE IV.—OH-58A TRANSMISSION GEAR TOOTH STRESS CYCLE COUNT

Component	Tooth load cycles per transmission input shaft revolutions	Tooth load cycles after 100 hr, transmission input speed 6060 rpm
Spiral bevel pinion	1.000	36.36×10^6
Spiral bevel gear	.268	9.73
Sun gear (meshing with three planets)	.631	22.94
Planet gear (each)	.162	5.89
Ring gear (meshing with three planets)	.172	6.25

gear bending stress was close to the upper limit at 262 kW. For contact stress at 262 kW, all components fell within the assumed allowable range. However, the bevel mesh contact stress was at the upper limit. Based on the AGMA stress analysis, a conclusion was drawn that 262 kW was the highest load at which to operate the OH-58A transmission to promote planetary pitting fatigue failures without causing tooth bending fatigue failures. Table IV gives the relative number of tooth load cycles for each component of the OH-58A transmission. Since the contact stress on the sun gear equals that on the planet gear, and since the number of load cycles on a sun gear tooth is nearly four times that of a planet tooth, sun gear pitting fatigue failures can be expected to occur significantly more often than planet gear fatigue failures.

At present, there is little agreement about the best method for evaluating spiral bevel gear scoring (ref. 13). Formulas for rating scoring resistance of bevel gear teeth are not included in AGMA standards. The method used to evaluate bevel gear scoring of the OH-58A transmission considered the flash temperature (defined as the total tooth surface temperature as it engages in mesh) compared to an assumed allowable value (ref. 14). Using this method, the flash temperature as a function of oil inlet temperature and transmission input power was estimated (fig. 8). For all conditions in figure 8, the predicted flash temperatures were below the assumed allowable for AISI 9310 gears (the material of the OH-58A spiral bevel gears) with MIL-L-23699 lubricant, indicating a low probability that scoring would occur.

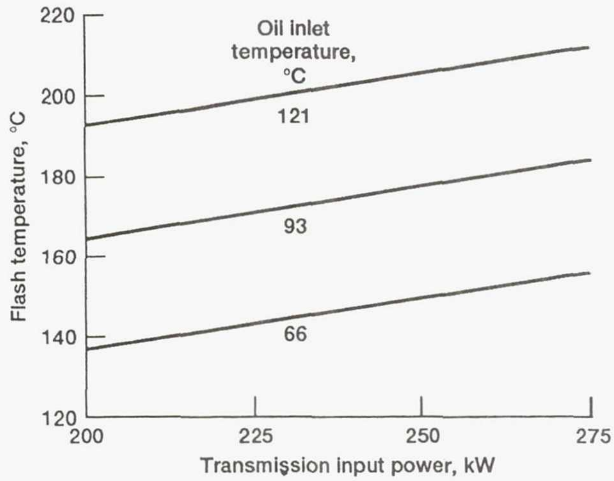


Figure 8.—OH-58A spiral bevel mesh flash temperature predictions. Allowable flash temperature to resist scoring = 257 °C for AISI 9310 gears with MIL-L-23699 lubricant (ref. 14).

Planet Bearing and Gear Analysis

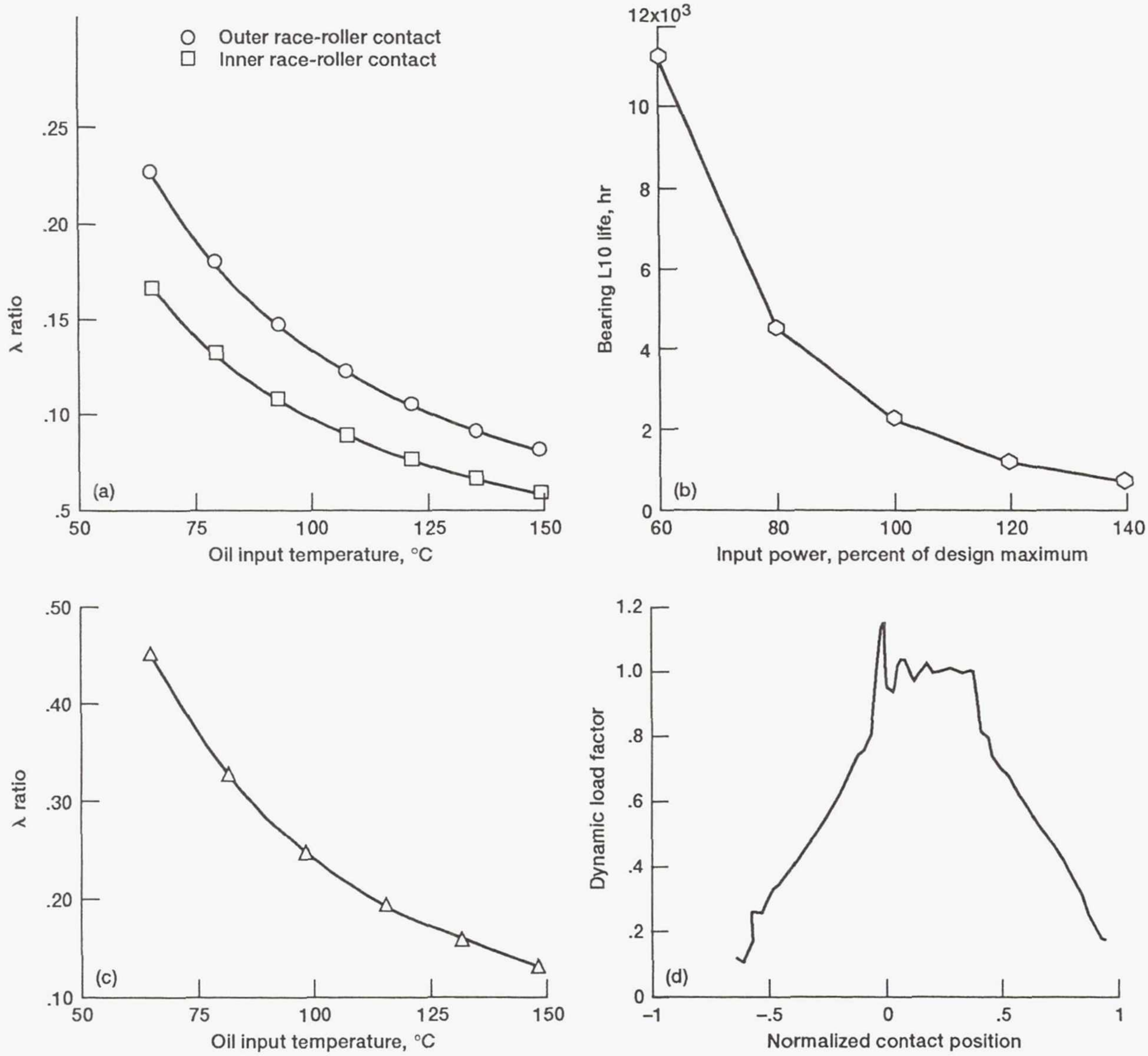
The OH-58A transmission's planet bearing was analyzed with the NASA/SKF computer program SPHERBEAN (ref. 15). Program inputs were bearing geometry, material properties, lubrication properties, and the operating conditions of speed, load, and temperature. Program outputs were roller kinematics, rollerraceway loads, stresses, film thicknesses, heat generation, and life. In all computer runs, typical properties for a MIL-L-23699 lubricant were used.

The predicted film thickness was extremely low, primarily because of the relatively low bearing speed (fig. 9(a)). At 79 °C (175 °F) oil inlet temperature, the calculated λ ratio (defined as the film thickness divided by the composite surface roughness) was 0.18 for one row of the outer race-roller contact and 0.13 for the inner race-roller contact. At these low λ values, a significant reduction in bearing life is predicted using American Society of Mechanical Engineers (ASME) design guides (ref. 16). In fact, the predicted λ values were lower than the lower limit of the ASME charts. As the bearing temperature increased, the predicted λ values decreased even further because of the lubricant's decreased viscosity. The predicted film thickness was not significantly affected by bearing load for the cases studied (60- to 140-percent of design maximum).

The effect of load on predicted bearing fatigue life is shown in figure 9(b). For these cases, the lubricant life adjustment factor based on the calculated λ ratio, as recommended by ASME (ref. 16), was constant at 0.21. From the predictions, the bearing was moderately loaded (maximum Hertz stress of 1.44×10^6 kPa (209 ksi) at 100-percent design maximum load). The predicted life was not affected by operating temperature for the cases studied (66 to 149 °C oil inlet). The reason was that the temperature effect on life was only considered in the lubrication life adjustment factor (which was constant at 0.21 for all cases, since the predicted λ ratios were lower than the lower limits of the ASME charts).

The OH-58A transmission sun-planet gear mesh was analyzed using NASA computer program TELSGE (refs. 17 and 18), which models a single spur gear mesh. Program inputs were gear geometry (including tooth profile relief), material properties, lubrication properties, and the operating conditions of speed, load, and temperature. Sun and planet gear speeds relative to a fixed planet carrier were used as inputs. Program outputs were dynamic loads, stresses, film thicknesses, bulk gear temperatures, and flash temperatures.

The predicted λ ratio was extremely low, about twice that of the planet bearing (fig. 9(c)). With increased temperature,



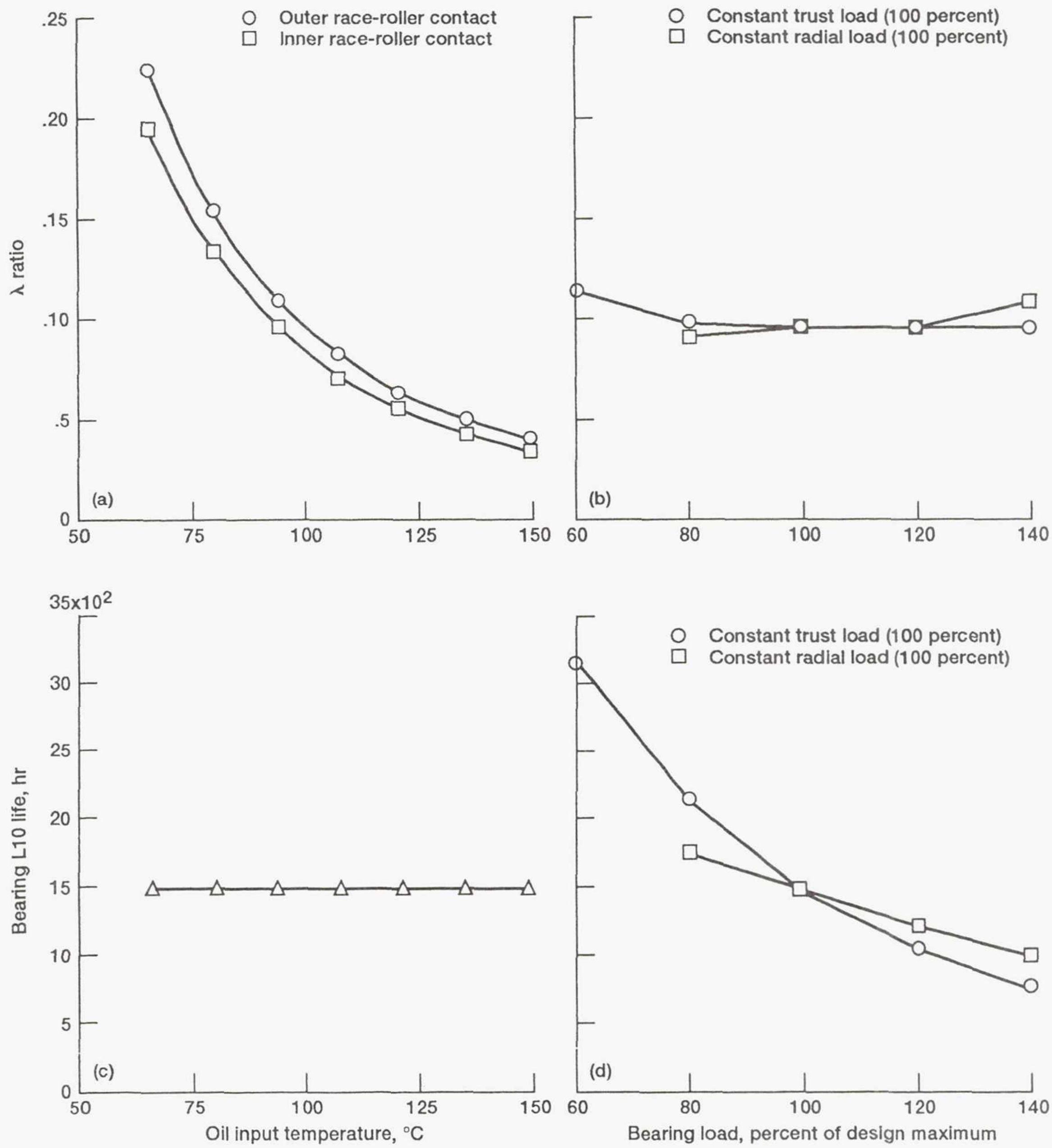
(a) Planet bearing single row film thickness at 100-percent design maximum load.
(b) Planet bearing life at 93 °C oil temperature.
(c) Planet gear film thickness at 100-percent design maximum load.
(d) Sun-planet gear dynamic load at 100-percent design maximum load.

Figure 9.—OH-58A transmission planet gear and bearing analysis.

the predictions portrayed trends similar to those of the planet bearing. The dynamic load factor (defined as the tooth dynamic load divided by the static load) as a function of normalized contact position (defined as the contact position divided by the base pitch) is shown in fig. 9(d). This curve was similar to a static load case indicating a low-speed condition and no dynamic loading. The low values predicted for film thicknesses and dynamic loads indicated the planet gear and bearing operate in a relatively low-speed regime. This would imply a greater probability of surface related problems, meaning the influence of the lubricant could be significant.

Mast-Shaft Ball Bearing Analysis

The OH-58A transmission mast-shaft ball bearing was analyzed using NASA/SKF computer program SHABERTH (ref. 19). Program inputs were bearing geometry, material properties, lubrication properties, and the operating conditions of speed, load, and temperature. Program outputs were ball kinematics, ball-raceway loads, stresses, film thicknesses, heat generation, and life. In all computer runs, properties for a MIL-L-23699 lubricant were used.



(a) Film thickness at 100-percent design maximum loads.
(b) Inner race-ball contact film thickness at 93 °C oil temperature.
(c) Bearing life at 100-percent design maximum loads.
(d) Bearing life at 93 °C oil temperature.

Figure 10.—OH-58A transmission mast-shaft ball bearing analysis.

The predicted film thickness was extremely low because of the low bearing speed (fig. 10(a)). At 79 °C (175 °F) oil inlet temperature, the calculated λ ratio was 0.15 for the outer race-ball contact and 0.13 for the inner race-ball contact. As with the planet bearing, the predicted λ values were lower than the lower limit of the ASME charts. As the bearing temperature increased, the predicted λ ratios decreased even further because of the lubricant's decreased viscosity. The predicted film thickness was not affected by bearing radial or thrust loads (fig. 10(b)).

The predicted fatigue life of the bearing was not affected by operating temperature for the cases studied (fig. 10(c)). For these cases, the ASME lubricant life adjustment factor was constant at 0.21. Increased radial load had a more significant effect on predicted bearing life than did increased thrust load (fig. 10(d)), because the life calculation was strongly influenced by the most heavily loaded ball-raceway contact. All the balls reacted equally to the thrust load, whereas only the most heavily loaded balls reacted to the radial load. The predictions indicated the bearing was heavily loaded (maximum Hertz stress of 2.98×10^6 kPa (432 ksi) at 100-percent design maximum load), but the life in hours was reasonable because of the low cycle count of the low-speed application. Overall, the low predicted λ ratios imply a greater probability of surface related problems such as micropitting. As with the planetary system, the effect of the lubricant on performance could be significant.

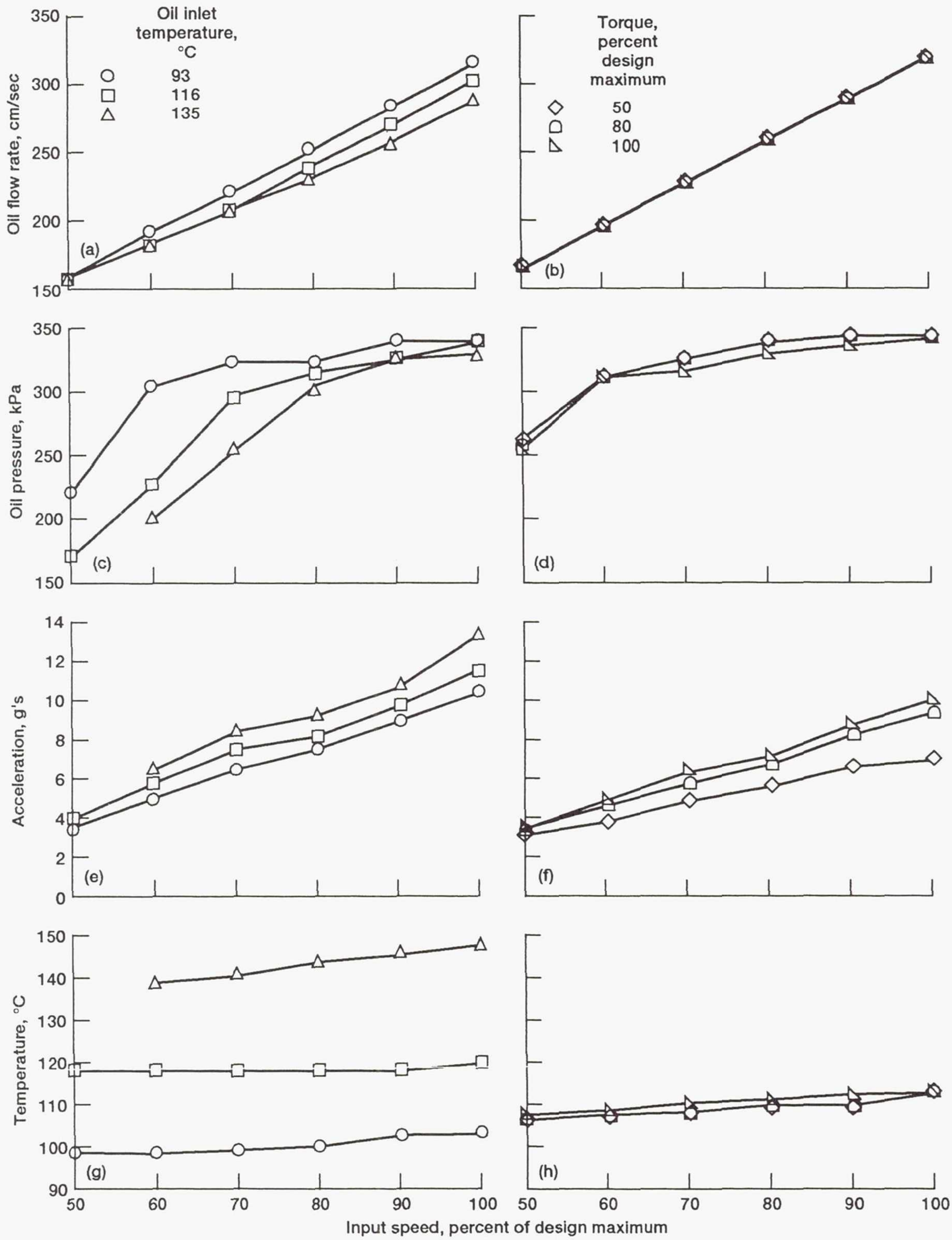
Appendix B
Results of Test 1 Parametric Studies

The OH–58A transmission speed, torque, and oil inlet temperature were varied to see their effect on transmission performance. The oil-flow rate increased linearly with speed (fig. 11(a)) since the bevel gear drives the oil pump. Increased temperature decreased flow slightly (fig. 11(a)), and transmission torque had no effect on it (fig. 11(b)). Oil pressure also increased with speed (since the bevel gear drives the pump) and decreased with temperature (fig. 11(c)). Transmission torque had little effect on pressure (fig. 11(d)). Generally, vibration increased with speed and temperature (fig. 11(e)). Vibration also increased with torque (fig. 11(f)). The vibration depicted came from an accelerometer mounted on the transmission housing near the ring gear spline. Vibration readings from accelerometers mounted at other locations on the transmission housing gave similar results. Some had peak amplitude responses at 90 percent of design maximum speed rather than at 100 percent.

The effect of speed on temperature was insignificant for most components. An exception was the spiral bevel pinion triplex ball bearing which showed a slight rise in temperature as speed increased (figs. 11(g) and (h)). Increased oil inlet temperature had little effect on component temperature other than raising the bulk oil temperature (fig. 11(g)). Also torque had little affect on component temperature (fig. 11(h)). For these tests, the oil inlet temperature was set at a desired value, and the oil-water heat exchanger maintained that temperature.

Decreased oil pressure had an effect on certain components (fig. 12). The planet bearings and the spiral bevel pinion triplex ball bearings were affected the most. Their temperatures increased noticeably with decreased oil pressure, but temperature of other components was not affected. Apparently, the transmission was designed with ample lubrication since a significant loss of oil pressure was required to noticeably affect component performance. Transmission mechanical efficiency decreased slightly when mast-shaft loading was introduced (fig. 13).

Efficiency also increased with torque. This was consistent with other reported studies on transmissions (refs. 20 and 21). Overall, the parametric studies gave valuable insight into the effect of operating conditions on transmission performance.



(a) Transmission oil-flow rate at 100-percent design maximum torque.
(b) Transmission oil-flow rate at 82 °C oil inlet temperature.
(c) Transmission oil pressure at 100-percent design maximum torque.
(d) Transmission oil pressure at 82 °C oil inlet temperature.
(e) Transmission housing vibration at 100-percent design maximum torque.
(f) Transmission housing vibration at 82 °C oil inlet temperature.
(g) Bevel pinion ball bearing inner race temperature at 100-percent torque.
(h) Bevel pinion ball bearing inner race temperature at 82 °C oil inlet.

Figure 11.—Results of test 1 parametric studies.

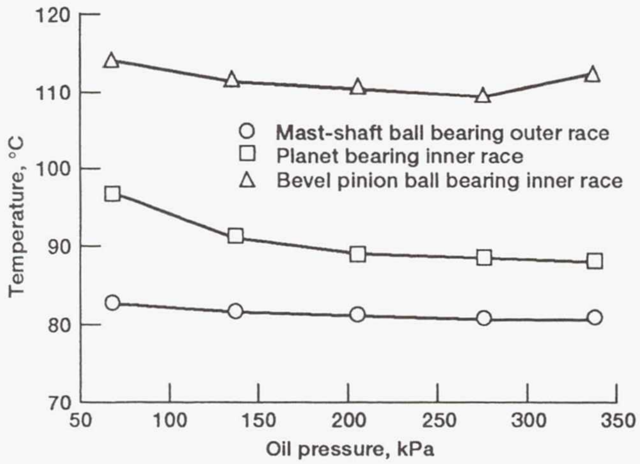


Figure 12.—Component temperatures as a function at 100-percent speed and torque, 82 °C oil inlet temperature.

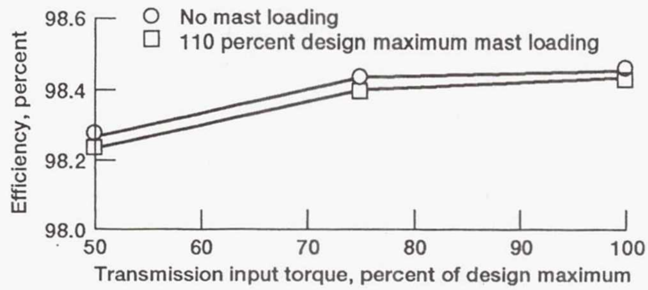


Figure 13.—Transmission efficiency as a function of torque at 100-percent speed, 82 °C oil inlet temperature.

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13. ABSTRACT (*Maximum 200 words*)

Experimental tests were performed on the OH-58A helicopter main rotor transmission in the NASA Lewis 500-hp Helicopter Transmission Test Stand. The testing was part of a joint Navy/NASA/Army lubrication program. The objective of the program was to develop a separate lubricant for gearboxes and demonstrate an improved performance in life and load-carrying capacity. The goal of the experiments was to develop a testing procedure to fail certain transmission components using a MIL-L-23699 base reference oil, then run identical tests with improved lubricants and demonstrate improved performance. The tests were directed at failing components that the Navy has had problems with due to marginal lubrication. These failures included mast shaft bearing micropitting, sun gear and planet bearing fatigue, and spiral bevel gear scoring. A variety of tests were performed and over 900 hours of total run time accumulated for these tests. Some success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. Only marginal success was achieved in producing mast shaft bearing micropitting and spiral bevel gear scoring.

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